

## Measurement Tools and Strategies to improve stability in Metal cutting

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### ABSTRACT

The prediction of appropriate cutting parameters to reduce cycle time and maintain product quality is major challenge in manufacturing industries. Regenerative chatter is a well known issue caused by the dynamic characteristics of tool, workpiece and spindle configuration resulting in self excited vibrations and unstable process output. In this paper CutPro is used to avoid chatter for different tool-holder settings and predict stable cutting parameters. The influences on the systems' dynamic characteristics are analysed using modal analysis of data from impact tests on various tool-holder combinations. The results show the influence of tool holder mass and length on structural flexibility and achievable predicted depths of cut.

**Index Terms** - Stability Lobes, Vibration Analysis, Metal Cutting

### INTRODUCTION

Automotive Industry Manufacturers pursuing for flexible and cheap ways to increase output and product quality. Stable processes guarantee reduced cycle times and excellent product features at low cost and minimum of waste. To realize proper outcome, Manufacturers looking for innovative solutions to observe, measure and improve production processes in line. A useful instrument for on-demand measuring of tooling machines, which has been established in the last years, is the Modal Analysis. The largest reduction in milling

quality occurs from a regenerative phenomenon called "Chatter Vibration" [1]. The vibration is the reason of poor surface and a loss of productivity. It produces more costs because of increasing tool and machine wear. Since the conception of modern manufacturing and CNC machining, several investigations have sought to deal with Chatter Vibration. A result of these researches is the Stability Lobe Diagram, which is one of the leading models to select optimum machine parameters. The approach in this paper is to improve process stability with Modal Analysis and Vibration testing. Taylor started a century ago with investigation to improve the process of cutting metal [1]. The first scientists, who developed a predictive model for stable process conditions and begin research in chatter vibration, was Tobias [2] and Tlustý [3]. They discovered that the main source of self-excitation, regeneration, is the dynamics of the machine tool-workpiece system and the feedback between the subsequent cuts on the same cutting surface. They used an orthogonal chatter model, which based on average directional milling coefficients and average number of teeth in cut. To describe the milling process accurately several parameters, as the rotating tool, multiple cutting teeth, periodical cutting forces or chip-load directions are missing in that model. Sridhar et al. [4] researched also the chatter stability. They achieved an elimination of the periodic and time delay terms in the stability model by using a systems state transition matrix. Minis et al. and Yanushevsky [5, 6] introduced a two-degree-freedom cutter model with point

contact. To predict stability limits they took the Nyquist criterion into account. Lee et al. [7, 8] replaced the time varying directional coefficients by a constant with the mean value method. The analytic approach from Altintas and Budak [9] is that the average value in the Fourier series expansion of the time varying coefficients was adopted. Many other researches use this approach. Later, Altintas and Budak [10, 11] also started researches, which include the harmonic terms. The outcome of the investigation was that the multi-frequency solution results are nearly the same as the in single frequency solution.

In this paper the feasibility of Modal Analysis and Vibration testing for an a priori predication of process stability and chatter avoidance is proven in an industrial environment. In this context frequency based approaches using impact testing and spindle acceleration measurements are taken into account. Sensible strategies for tooling machine measurement and simulation of stability lobes are evaluated with respect to contemporary production issues. Trails to calculate stability lobes by fitting them under consideration of a few cutting trials are accomplished as well. This requires less time to acquire data and predict stable cutting parameters.

## ANALYTICAL MODEL

The following chapter introduces the physical and mathematical background to understand the subsequent results. In this instance the stability model based on Altintas und Budak [9] will be summarized.

Fig. 1 shows the differential forces in the milling process. Milling cutter and workpiece have two orthogonal modal directions allowing the cutter to move in the x-y plane.

The periodic milling forces induce dynamic displacements, which cause a wavy surface on the workpiece. Every tooth in cut leaves a wave on the surface, which is interleaved by a likely shaped path of the following tooth. The phase shift between two following teeth

in cut causes a modulation of chip thickness inducing subsequent vibration into the system.

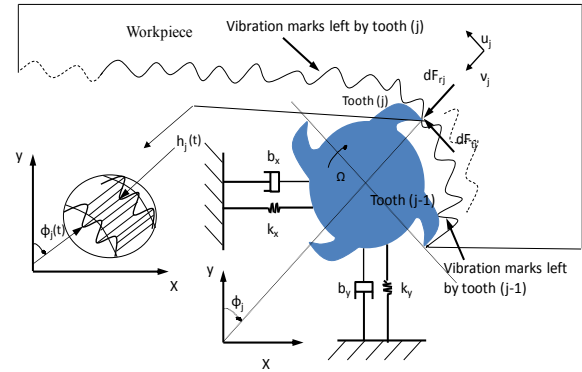


Fig. 1 - Differential forces in the milling process [9, 10, 11]

The phase shift is summarized in the modulated chip thickness which can be express as:

$$h_j(\phi) = f_t \sin \phi_j + \begin{pmatrix} 0 \\ j_c \end{pmatrix} - \begin{pmatrix} 0 \\ j_w \end{pmatrix} - \begin{pmatrix} 0 \\ j_c \end{pmatrix} + \begin{pmatrix} 0 \\ j_w \end{pmatrix}, \quad (1)$$

$f_t$  is the feed per tooth.  $\phi_j = (j-1) \phi_p + \Omega t$  ( $\Omega$  rotational speed) is the angular immersion of the tooth (j) for a cutter with constant pitch angle  $\phi_p = 2\pi/N$  (N number of teeth).  $\begin{pmatrix} 0 \\ j_c \end{pmatrix}$  and  $\begin{pmatrix} 0 \\ j_w \end{pmatrix}$  are the dynamic displacements in the chip thickness direction due of the cutter and workpiece at the current and previous tooth passes.

$$v_{jp} = -x_p \sin \phi_j - y_p \cos \phi_j \quad (p = c, w), \quad (2)$$

$j$  is the angular position. c and w stands for cutter and workpiece.  $x_p$  and  $y_p$  are the dynamic displacements in global axis. After neglecting the static part in (1), which is can be eliminated in the chatter analysis, is the dynamic chip thickness defined as:

$$h_j(\phi) = [\Delta x \sin \phi_j + \Delta y \cos \phi_j], \quad (3)$$

where

$$\begin{aligned} \Delta x &= (x_c - x_c^0) - (x_w - x_w^0), \\ \Delta y &= (y_c - y_c^0) - (y_w - y_w^0), \end{aligned} \quad (4)$$

The dynamic cutting force on a tooth in tangential ( $F_{tj}$ ) and radial ( $F_{rj}$ ) direction is defined as:

$$F_{t_j}(\phi) = K_t a h_j(\phi); F_{r_j}(\phi) = K_r F_{t_j}(\phi) \quad (5)$$

The axial depth of cut is defined as  $a$ .  $K_t$  and  $K_r$  are the cutting force coefficients. In global directions and after substitution  $h_j$  the cutting forces can be expressed as:

$$\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \frac{1}{2} a K_t \begin{bmatrix} a_{xx} & a_{xy} \\ a_{yx} & a_{yy} \end{bmatrix} \begin{Bmatrix} \Delta x \\ \Delta y \end{Bmatrix}, \quad (6)$$

where  $a_{xy}$  are the directional coefficients and depend on the angular position. This makes Eq. (6) time varying:

$$\{F(t)\} = \frac{1}{2} a K_t [A(t)] \{\Delta(t)\} \quad (7)$$

$[A(t)]$  is periodic at the tooth passing frequency  $\omega = N\Omega$ . For the solution of periodic systems Fourier series expansion is applied. Altintas and Budak [10, 11] revealed that only the average term in the fourier series expansion of  $[A(t)]$  is sufficient for the solution. In that case the directional coefficients of  $[A(t)]$  are shaped as

$$\begin{aligned} a_{xx} &= \frac{1}{2} [\cos 2\phi - 2K_r\phi + K_r \sin 2\phi]_{\phi_{st}}^{\phi_{ex}} \\ a_{xy} &= \frac{1}{2} [-\sin 2\phi - 2\phi + K_r \cos 2\phi]_{\phi_{st}}^{\phi_{ex}} \\ a_{yx} &= \frac{1}{2} [-\sin 2\phi + 2\phi + K_r \cos 2\phi]_{\phi_{st}}^{\phi_{ex}} \\ a_{yy} &= \frac{1}{2} [-\cos 2\phi - 2K_r\phi - K_r \sin 2\phi]_{\phi_{st}}^{\phi_{ex}} \end{aligned} \quad (8)$$

[10, 11]. Considering Eq. (8), Eq. (7) reduces to:

$$\{F(t)\} = \frac{1}{2} a K_t [A_0] \{\Delta(t)\} \quad (9)$$

$[A_0]$  is a time invariant cutting coefficient matrix.

The dynamic displacement vector in Eq. (9) can be solved from the FRF (frequency response function) and the dynamic forces. By replacing the response and the delay terms the following term is obtained:

$$\{F\} e^{i\omega_c t} = \frac{1}{2} a K_t (1 - e^{-i\omega_c T}) [A_0] [G(i\omega_c)] \{F\} e^{i\omega_c t} \quad (10)$$

$\{F\}$  constitutes the amplitude of the dynamic cutting force  $\{F(t)\}$  and  $[G(i\omega_c)]$  is the

transfer function matrix, which identified at cutter – workpiece contact zone,

$$[G(i\omega)] = \begin{bmatrix} G_{xx}(i\omega) & G_{xy}(i\omega) \\ G_{yx}(i\omega) & G_{yy}(i\omega) \end{bmatrix} \quad (11)$$

where  $G_{xx}(i\omega)$  and  $G_{yy}(i\omega)$  are direct transfer functions in  $x$  and  $y$  directions.  $G_{yx}(i\omega)$  and  $G_{xy}(i\omega)$  are the cross transfer functions.

Eq. (10) has a non-trivial solution only its determinant is zero,

$$\det[I + \Lambda[G_0(i\omega_c)]] = 0, \quad (12)$$

$[I]$  is the unit matrix and Eq. (13) defines the oriented transfer matrix.

$$[G_0] = [A_0][G] \quad (13)$$

The eigenvalue  $\Lambda$  can easily be calculated from Eq. (12) numerically and is obtained as:

$$\Lambda = -\frac{N}{4\pi} K_t a (1 - e^{-i\omega_c T}) \quad (14)$$

If two orthogonal degrees of freedom in feed ( $X$ ) and normal ( $Y$ ) directions are considered ( $G_{yx}$  and  $G_{xy}$  are neglected) an analytical solution is possible:

$$\Lambda = -\frac{1}{2a_0} \left( a_1 \pm \sqrt{a_1^2 - 4a_0} \right) \quad (15)$$

where

$$\begin{aligned} a_0 &= G_{xx}(i\omega_c) G_{yy}(i\omega_c) (\alpha_{xx} \alpha_{yy} - \alpha_{xy} \alpha_{yx}) \\ a_1 &= \alpha_{xx} G_{xx}(i\omega_c) + \alpha_{yy} G_{yy}(i\omega_c). \end{aligned} \quad (16)$$

The numerical solution of Eq. (12) will give eigenvalues with complex and real parts,  $\Lambda = \Lambda_R + i\Lambda_I$ . The axial depth of cut ( $a$ ) has a real value. After substituting the eigenvalue and  $e^{-i\omega_c T} = \cos \omega_c T - i \sin \omega_c T$  in Eq. (14), the complex part of the equation vanishes:

$$K = \frac{\Lambda_I}{\Lambda_R} = \frac{\sin \omega_c T}{1 - \cos \omega_c T}. \quad (17)$$

This equation can be used to obtain a relation between the chatter frequency and the spindle speed [9, 10, 11]:

$$\begin{aligned}
\omega_c T &= \varepsilon + 2k\pi \\
\varepsilon &= \pi - 2\psi \\
\psi &= \tan^{-1} K \\
n &= \frac{60}{NT}
\end{aligned} \tag{18}$$

Where  $\varepsilon$  is the phase difference between the inner and outer undulations,  $k$  is an integer corresponding to the number of vibration waves within a tooth period and  $n$  is the spindle speed (rpm). After eliminating the imaginary part in (14) the final expression for chatter free axial depth of cut is:

$$a_{lim} = -\frac{2\pi\Lambda_R}{NK_t}(1 + K^2) \tag{19}$$

Hence for a given chatter frequency  $\omega_c$  the eigenvalues are obtained from (12), which permit the critical depth of cut to be computed (19) and finally the spindle speed using (16) for the different number of vibration waves,  $k$ . This procedure is frequent for various chatter frequencies around the structures dominant modes. The stability lobe diagram for a milling system is obtained. The diagrams will be used to locate the maximum chatter free cutting depths and spindle speeds.

## EXPERIMENTS

### 1.1 Measurement setup

To identify how Tool-Mass und -Length affects transfer functions, analytical stability Lobes impact tests were conducted on two evenly matched cutters mounted in different tool holder extensions.

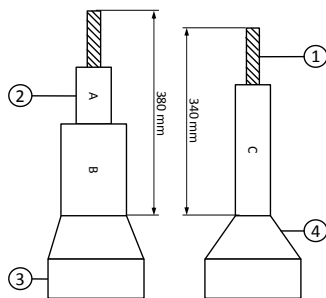


Fig. 2 – Tool holder geometry: 1 cutting tool, 2 tool holder extension, 3 Spindle, 4 clamping

The first configuration by using extension C achieves an overall tool length of 340 mm

and mass of 7.5 kg. To reach an aggregated tool length of 380 mm extension A is mounted on B with a total mass of 12 kg as seen in Fig. 2. Additional information on tool and holder extension parameters is shown in Table 1.

Table 1. Cutting tool and extension parameters

Tool parameters			
Cutter type	End mill		
Diameter (mm)	40	Helix angle	40°
Tool length (mm)	140	Relief angle	7°
Evenly spaced Flutes	3	Rake angle	14°
Extension parameters			
	A	B	C
Diameter (mm)	63	100	63
Length (mm)	100	140	200

The impact tests were performed using a Kistler type 9722A500 Impulse hammer and a Kistler 8776A50 type accelerometer. To capture signal data a National Instruments 9234 Data acquisition card is used. The Acquisition device is connected to a storage Device and Signals are processed by simulation tool CutPro. In order to identify the Static flexibility of any given tool, -holder, spindle-assembly the tool tip gets excited by the impact of an impulse hammer. The inducted vibrations are measured and processed by an accelerometer as shown in Fig. 3.

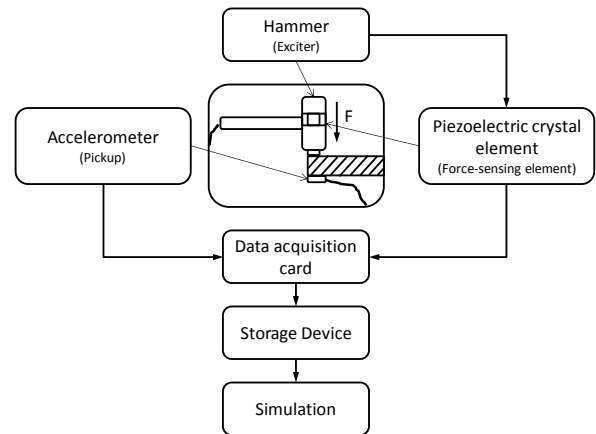


Fig. 3 – Measurement chain

### 1.2 Results

The frequency response function as seen in equation 10 equals the ratio of the measured output over the input.

$$[G(i\omega)] = \frac{\{Q(\omega)\}}{\{F(\omega)\}} \tag{19}$$

where  $\{Q(\omega)\}$  is the vibration vector at a present time [12].

$$\{Q(\omega)\} = \{x(t) \ y(t)\}^T \quad (20)$$

And  $\{F(\omega)\}$  equals the vector of the applied impact Forces. The FRF's processed by CutPro depending on the obtained acceleration und Force data, transferred by the impulse hammer, are used to compare the different Tool holders concerning static structural Flexibility as well as the simulated Stability lobes.

Figure Fig. 4 shows the real and imaginary part of the transfer function obtained in feed direction of the tested tool-holder combinations.

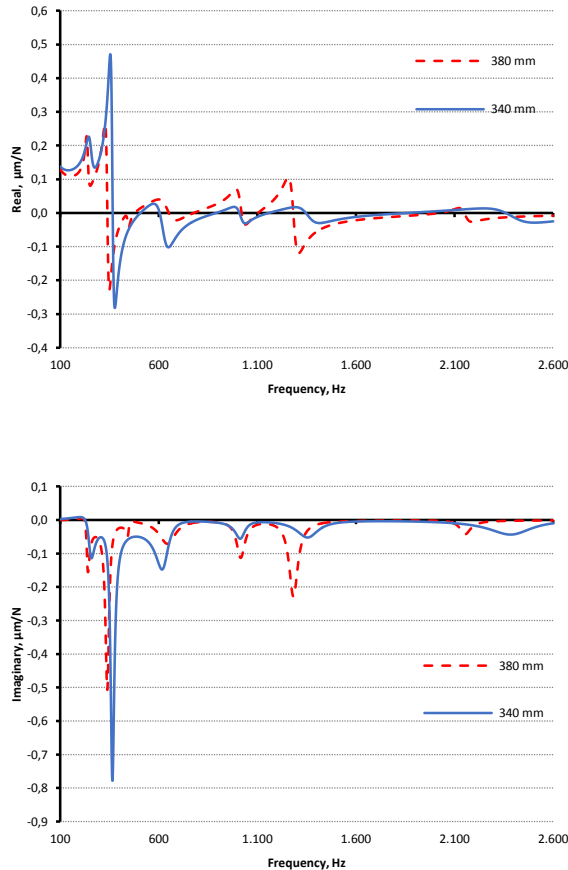


Fig. 4 – Real- and imaginary part

The slight difference in Amplitude of Real- and Imaginary-part points at a higher overall stiffness in the 380 mm configuration. The maximum negative peak of the Real-part is in both cases around a frequency of 274 Hz.

At this frequency the system will tend to chatter most likely. Figure Fig. 5 shows the Structural flexibility (Modulus of FRF in  $\mu\text{m/N}$ ) and the Phase shift (Phase in  $^\circ$ ).

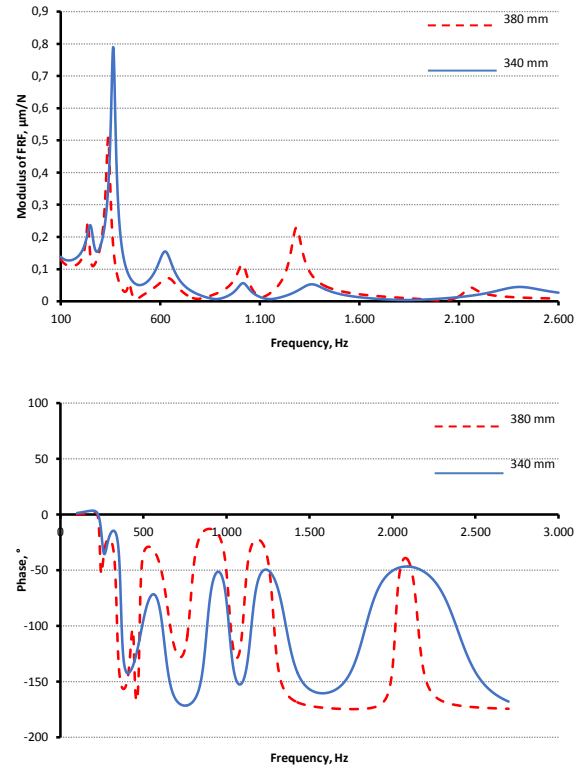


Fig. 5 – Structural flexibility and Phase response

The highest peak of both curves is at a frequency of 360 Hz differing in Amplitude about  $0,3 \mu\text{m/N}$ . The Phase response differs in Phase and Frequency. The highest phase shift is reached by the 380 mm configuration at a frequency of 1820 Hz.

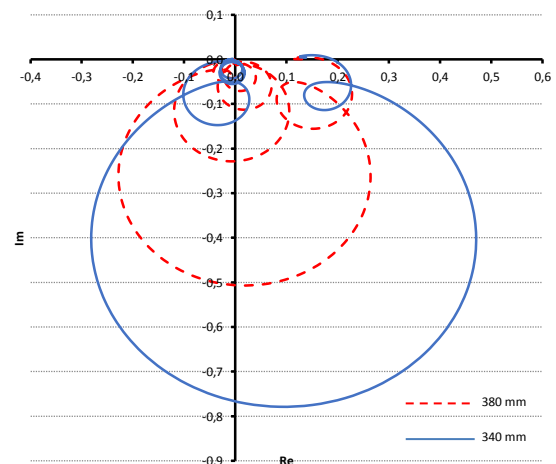


Fig. 6 – Nyquist Plot

Fig. 6 confirms the higher static structural flexibility of the 340 mm configuration making this system act softer in comparison to the 380 mm configuration. The lower stiffness can be reduced on the lack mass in shape of the missing tool holder B. The highest overall negative value noticed in the Nyquist Plot shows a higher chatter affinity of the 340 mm configuration.

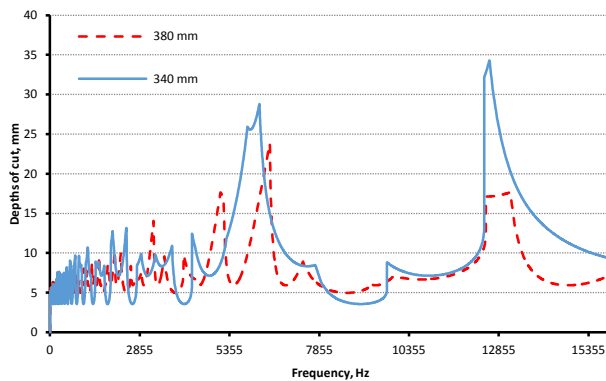


Fig. 7 – Predicted stability lobes

Considering the simulated stability lobes (Fig. 7), the additional dominant peak noticed at 1280 Hz in the 380 mm configuration, seen in Fig. 5, might be correlated with the additional sectioning between holder extension A and B. This is having a negative influence on the reachable depths of cut as seen in Fig. 7.

An extract of the simulation parameters applied in CutPro to predict stable cutting parameters are summarized in Table 2.

Table 2. Extract of simulation parameters

Analytical Stability Lobes	
Edge radius (mm)	0.02
Flank wear (mm)	0.05
Spindle direction	Clockwise
Milling mode	Up-milling
Cutting parameters	
Feedrate (mm/flute)	0.09
Radial width of cut (mm)	1

## SUMMARY AND OUTLOOK

The conducted measurements show slight differences in the prediction of stability lobes due to the variation of tool-holder mass and length. It is shown that even though the 380 mm configuration is stiffer, it reaches lower

depth of cut over the available spindle speed spectrum. An experimental validation of the predicted cutting parameters has to be done with respect to trade-offs in terms of output loss, cost per part and Quality reduction in series production.

In future tasks the dynamic spindle-holder-tool flexibility should be taken into account to compare results and evaluate the influence on cutting stability. Moreover additional Measurements should be conducted to identify further influencing factors on cutting stability in industrial environments. The analysis of additional tool-holder configurations can provide useful data for prospective tool layouts as well as improvements for an effective construction.

## ACKNOWLEDGMENT

The Author would like to thank the Manufacturing Automation Laboratory (Yusuf Altintas from University of British Columbia and Martin Herrmann) for technical Support.

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